Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

1972

Performance Improvement and Dimensional Commonization With a Compressor Simulation

D. A. Coates
Whirlpool Corporation

PERFORMANCE IMPROVEMENT AND DIMENSIONAL COMMONIZATION WITH A COMPRESSOR SIMULATION

Donald A. Coates, Research Engineer
Whirlpool Corporation, Benton Harbor, Michigan

INTRODUCTION

The objectives of this work were to eliminate dimensional differences and to improve performance of a family of compressors through use of a compressor simulation. After consideration of the manufacturing flexibilities, seven parameters were studied for commonality and an additional five were studied for performance improvement.

There are six compressor capacity (Btu/Hr) sizes in the family. For simplicity they will be referred to as compressors A through F. The capacity of compressor F is approximately twice that of compressor A. The other compressors are approximately evenly distributed between them.

Figure 1 shows a schematic of a typical compressor used in this study. The compressor shown is one of the hermetically encased types used in refrigerators, freezers, and air conditioners. This design has a single lobe (the volume for gas resembles a lobe or crescent) which is divided into three compartments by the two vanes and the minimum clearance.

The operation of the compressor is quite simple in nature. The cylinder of the rotary vane compressor is stationary while the rotor turns about its geometric center. The rotor rotation and the eccentricity of the cylinder wall with respect to the rotor center cause the volume enclosed between any two vanes or vane and minimum clearance to vary from a maximum to a minimum during each revolution. This action causes gas to be drawn in at the suction ports, compressed, and discharged at the discharge port. The discharged gas is prevented from returning to a compartment by the flexible reed which serves as a check valve. Also, the transfer slot carries over the oil and residual gas to the following chamber, rather than trying to squeeze them passed the minimum clearance to the suction side. This latter action, if allowed, could give rise to damaging forces and a gas carryover that would degrade the volumetric efficiency. Lastly, the vanes are held against the cylinder by the discharge pressure acting on the underside (part of the vane closest to the rotor center) of the vane.

The justifications for performance improvement are quite classical and thus do not need reiterating, however, those for commonality may not be as clear. In production it is desirable to have as many common parts in a family of compressors as possible. This increases productivity since there are fewer tooling changes; there is less scheduling, less supervisory work, and less downtime required to change to a different capacity pump. This leads to decreased manufacturing costs and may make future automation more feasible. As a further result of employing commonization, the inventory of parts and drawings is reduced.

Performance improvement and commonization studies can be carried out with a simulation at the inception of a design. The experimental method makes such studies difficult if not impossible.

Disadvantages of commonization are that the performance of some pumps may be less than and physical size may be larger than a "custom designed pump" (locally optimum). However, for this case, the results of this study will show these disadvantages to be insignificant.

The principal tool for this study is a computer simulation model of the rotary compressor. The model was derived from work done by Wambsganss and Cohen (1) on reciprocating compressors. There are many other fine models available; a representative group is listed in the references (2-12). The simulation model is based on a mathematical model which represents the thermodynamic processes in the compressor that significantly affect the performance.

The simulation model is programmed in Fortran and solved on a digital computer. Solution results include COP (coefficient of performance, Btu/(watt-Hr)), capacity (Btu/Hr), energy input (Watts), and many other performance quantities.

The study applied the simulation model to the A and F capacity compressors. They are the smallest and largest members in the compressor family. Plots of the coefficient of performance (COP) versus a design parameter such as discharge port diameter were made to study the sensitivity (slope) of the COP at different values of the parameter.

Typical curves for compressor sizes A and F are shown in Figures 2a, 2b, and 2c. The basic procedure was to find a single value of a parameter that gave on the average the highest overall performance for both the smallest and largest compressors in the family. It was then assumed this value would give on the average the highest overall performance for the whole family of compressors. Previous experience with the simulation showed that the character changes between the COP-parameter curves for different capacity pumps were in one direction. Thus only using the bracketing capacities is valid and most expedient with the present simulation.

PROCEDURE

The procedure begins with COP-parameter curves generated for the smallest and largest capacity compressors in the family. Figures 2a, 2b, and 2c show schematically some of the types of curves found in this study. For the case in 2a, a parameter value close to the peak performance for the A and F capacity pumps appears to give the best overall value; see the solid vertical line in Figure 2a. The curves shown in 2b indicate that the parameter should be made as small as possible if not eliminated. In 2c the largest practical value of the parameter should be used. A criterion other than performance will be the limiting factor in this case, e.g., the discharge port cannot be made larger than the available material.

Parameter values for the designs existing at the beginning of this study are designated by dashed vertical lines as is shown in Figure 2a. To avoid confusion, the dashed lines will only intersect the curves for which they apply.

Performance improvement is carried out in the same manner as commonization. The value of each parameter for this part of the study should be already common among the members of the compressor family. Thus the dashed vertical lines for the performance study like those in Figure 2 will be coincident. In a similar manner to the commonization procedure each parameter is varied independently to find an optimum performance. No coupled parameter variations are made. The differences between a coupled and uncoupled optimum are felt to be insignificant and thus not to warrant a more sophisticated optimization method for this particular study.

Once COP plots are made for all parameters which lend themselves to commonization and performance optimization, a composite set of parameter values is used in the simulation and run for each member of the compressor family. Figure 2d shows that for most pumps the new performance may deviate from the initial values; in some cases it may increase slightly and in other cases it may decrease slightly. The facts which are most significant are that the compressors in this family will now have a maximum of common dimensions and performance.

RESULTS

At the beginning of the study, each of the seven parameters in Table 1 were uncommon among the existing capacities of the compressor family. Table 1 shows the degree of parameter variation throughout the family before the study and the final values after commonization.

TABLE 1

Normalized Parameter Values*
Before and After Commonization

	Before			
Capacity Ratios	1 (A)	1.16 (B)	1.44 (C)	1.67 (D)
Normalized Parameter				
$Parameter_1$	1.0	1.0	1.38	1.38
$Parameter_2$	1.0	1.0	.75	.75
$Parameter_3$	1.0	1.0	2.23	2.23
$Parameter_4$	1.0	1.0	1.05	1.05
$Parameter_5$	1.0	1.1	1.24	1.36
$Parameter_6$	1.0	1.0	1.16	1.16
$Parameter_7$	1.0	1.0	1.55	1.55

*All values normalized with respect to the parameter values of compressor capacity A.

TABLE 1 (Continued)

	Before (Continued)		After
Capacity Ratios	1.84 (E)	2.16 (F)	
Normalized Parameter			Common Value
$Parameter_1$	1.38	1.54	1.68
$Parameter_2$	.75	.75	eliminated
$Parameter_3$	2.23	2.23	eliminated
$Parameter_4$	1.05	1.05	1.05
$Parameter_5$	1.42	1.51	1.51
$Parameter_6$	1.16	1.16	1.0
$Parameter_7$	1.55	1.55	1.70

The seven parameters studied are geometrically identified in Figure 3. RADHHH is the radius of the short transfer slot which can be seen in View A-A of Figure 3 with a height XGG. XL2 establishes the port length. The length of the pressure relief valve port is determined by XL6. DP or DP2 is the discharge port diameter. XL13, shown in View A-A, is the distance between the outside edges of the long transfer slots. Finally, XL8 is the chord length of the long transfer slot and is shown in View A-A of Figure 3 with height XG.(XG was not a parameter studied for commonality).

The variations for the five additional parameters that were studied for performance improvement are shown in Table 2.

TABLE 2

Normalized Parameter Values Before and After Optimization Study with Simulation

Normalized Parameter	Before	After
$Parameter_8$	1.0	1.0
$Parameter_9$	1.0	1.6
$Parameter_{10}$	1.0	eliminated
$Parameter_{11}$	1.0	1.0
$Parameter_{12}$	1.0	.375

The five additional parameters studied are identified in Figure 3. The radius of curvature of the discharge valve is RADCV. It determines the maximum height to which the valve can lift over the port. XHHH is the length of the short transfer slot. RADM is the radius of curvature of the long transfer slot. It is shown in View A-A with a height XG. Finally, XL1 locates the discharge port center.

To gain confidence that the compressors have been properly modeled and to provide a means for comparing simulation results "After Commonization", the simulation was run for the existing designs and compared to existing experimental results. Figure 4 shows these results. (In Figures 4 and 5 the A capacity pump corresponds to the smallest capacity ratio and subsequent alphabetic letters to correspondingly larger ratios. Table 1 also gives the equivalence between the ratios and alphabetical letters.) The maximum difference between the experimental and theoretical results is less than three percent. Each experimental point plotted is the averaged result from several identical but distinct compressors. For the C and E capacity compressors, the experimental compressors did not have all parameters completely identical to the ones used in the simulation. Therefore, the results for these two capacities are expected to be slightly different. Since the simulation validity has been established in previous applications, this phase was in essence mainly checking the input data.

DISCUSSION OF RESULTS

Using the values of the parameters listed in Tables 1 and 2 under "After" for all six pumps from the A to F capacity, the triangular points shown in Figure 5 were produced. Instead of losing performance in some areas and gaining in others, there has been an overall increase of over five percent.

The number of remaining uncommon parameters may be a minimum. The vane length, cylinder diameter and motor size are dependent on the capacity of the pump. The cooling capacity does not appear to be a theoretically feasible characteristic to commonize due to increased system costs and possibly lower performance of high flow rate designs.

The technique employed in this study shows that the D capacity compressor is the most efficient one of the members of this compressor family.

Furthermore, if a smaller size pump were used in this procedure along with the A capacity pump instead of the F capacity pump, the peak performance could be shifted to a lower capacity, say the B capacity.

Judging from the results in Figure 5 it appears that this method is very worthwhile in compressor design. Especially since the parameters listed in Table 1 have all been successfully commonized and those that characterize the short transfer slot RADHHH, XGG, and XHHH have been eliminated.

CONCLUSIONS

Based on the commonization and performance improvement study performed with the simulation, the following has been demonstrated:

a. Seven parameters were commonized (see Table 1 and context for description) that originally varied throughout the compressor family. Two parameters were eliminated from the design in this part of the study.
b. The performance was further improved by five percent for the entire compressor family. An additional parameter was eliminated from the design by the performance phase.
c. The comparisons made with experimental results at the start of the study and during the study show good agreement. These further verify that the simulation gives valid performance predictions.
d. The technique required a small number of computer simulations. This is mainly because only two compressors had to be used in the investigation. The others were used to evaluate the results at the end of the study.

This approach can be used as a design tool which could substantially improve new compressor performance and manufacturability before release to manufacturing. This could speed up the design process and reduce design costs.

ACKNOWLEDGEMENTS

The author gives thanks to those individuals in Whirlpool Corporation whose foresight and perseverance have made this work possible. He also thanks the staff of the Ray W. Herrick Laboratories of Purdue University for its always rewarding association. Finally, he thanks his wife, Pat, who in no small way helps him get the job done.

REFERENCES

1. M. Wambsganss, and R. Cohen, Simulation of Reciprocating Compressors with Automatic Reed Valves, Part I: Theory and Simulation; Part II: Experiments and Evaluation, Proceedings of the XII International Congress of Refrigeration, Madrid, Spain, August 30-September 6, 1967, Paper 3.06.

2. M.J. Stevenson, A Computer Simulation of a Rotary-Vane Compressor, M.S.M.E. Thesis, Mechanical Engineering Department, Purdue University, August, 1969.

3. G.W. Gatecliff, A Digital Simulation of a Reciprocating Hermetic Compressor Including Comparisons with Experiment, Ph.D. dissertation, University of Michigan, 1969.

4. W. Brunner, Simulation of a Reciprocating Compressor on an Electronic Analog Computer, presented at ASME Winter Annual Meeting, 1949.

5. M. Costagliola, The Theory of Spring-Loaded Valves for Reciprocating Compressors, ASME Transactions: Journal of Applied Mechanics, Vol. 72, Dec. 1950, pp 415-420.

6. J.D. Haseltine, Part-Load Performance of Reciprocating Compressors, M.S.M.E. Thesis, Mechanical Engineering Dept., Purdue University, 1970.

7. B.D. Kotalik, Computer Simulation of a Five Horsepower High Speed Reciprocating Compressor, M.S.M.E. Thesis, Mechanical Engineering Dept., Purdue University, 1969.

8. J.F.T. MacLaren and S.V. Kerr, Valve Behavior in a Small Refrigerating Compressor Using a Digital Computer,

Journal of Refrigeration, Volume 11, No. 6, June 1968.

9. E.B. Qvale, The Development of a Mathematical Model for the Study of Rotary-Vane Compressor, ASHRAE Transactions, Vol. 77, Part I, 1971.

10. D.D. Schwerzler, Mathematical Modeling of a Multiple Cylinder Refrigeration Compressor, Ph.D. dissertation, Mechanical Engineering Dept., Purdue University, June 1971.

11. E.M. White, Application of Mathematical Model to High Speed Reciprocating Compressors, M.S.M.E. Thesis, Mechanical Engineering Dept., Purdue University, 1969.

12. D.A. Coates, Design and Digital Computer Simulation of a Reciprocating Free Piston Electrodynamic Gas Compressor, M.S.M.E. Thesis, Mechanical Engineering Dept., Purdue University, August, 1966.

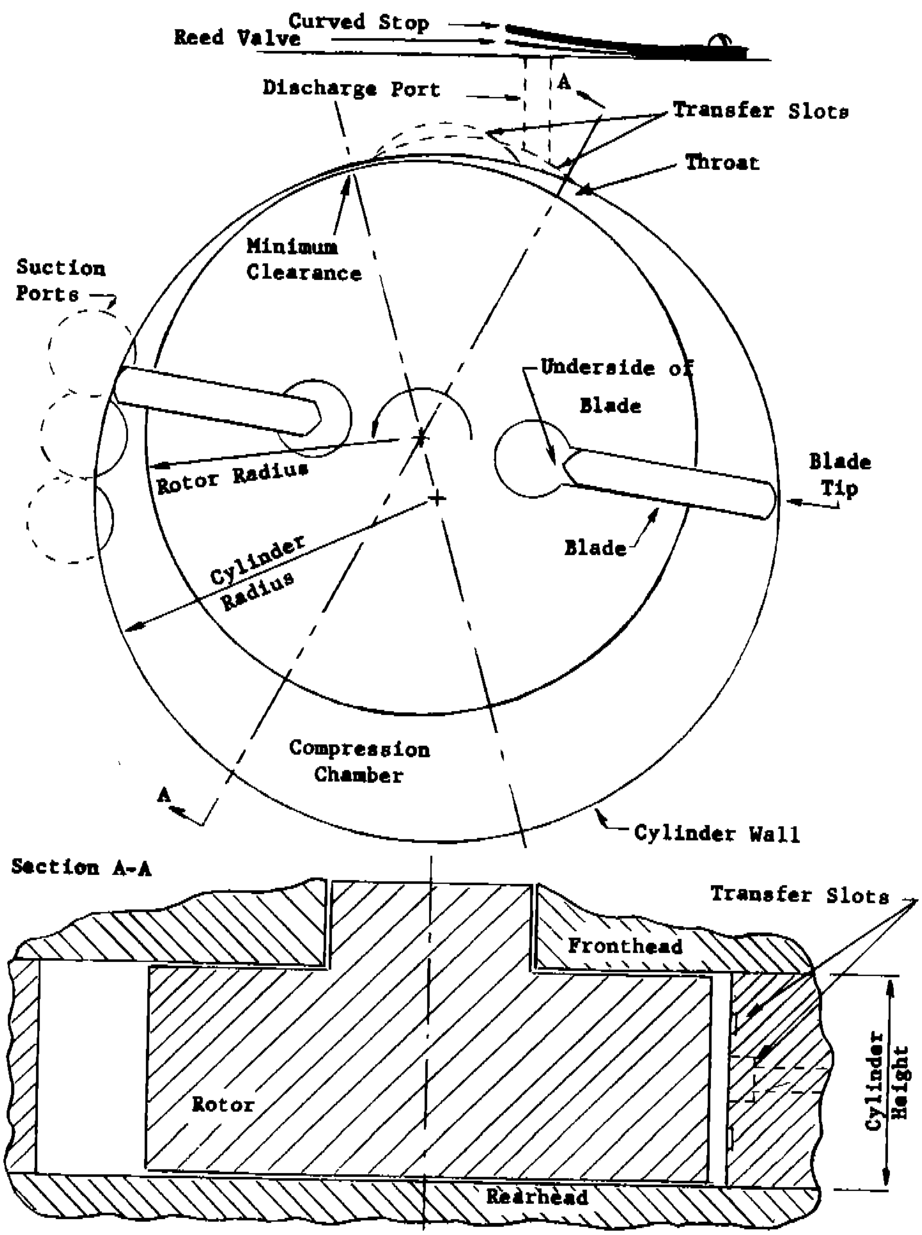


FIGURE 1 - BASIC PARTS OF A ROTARY VANE COMPRESSOR

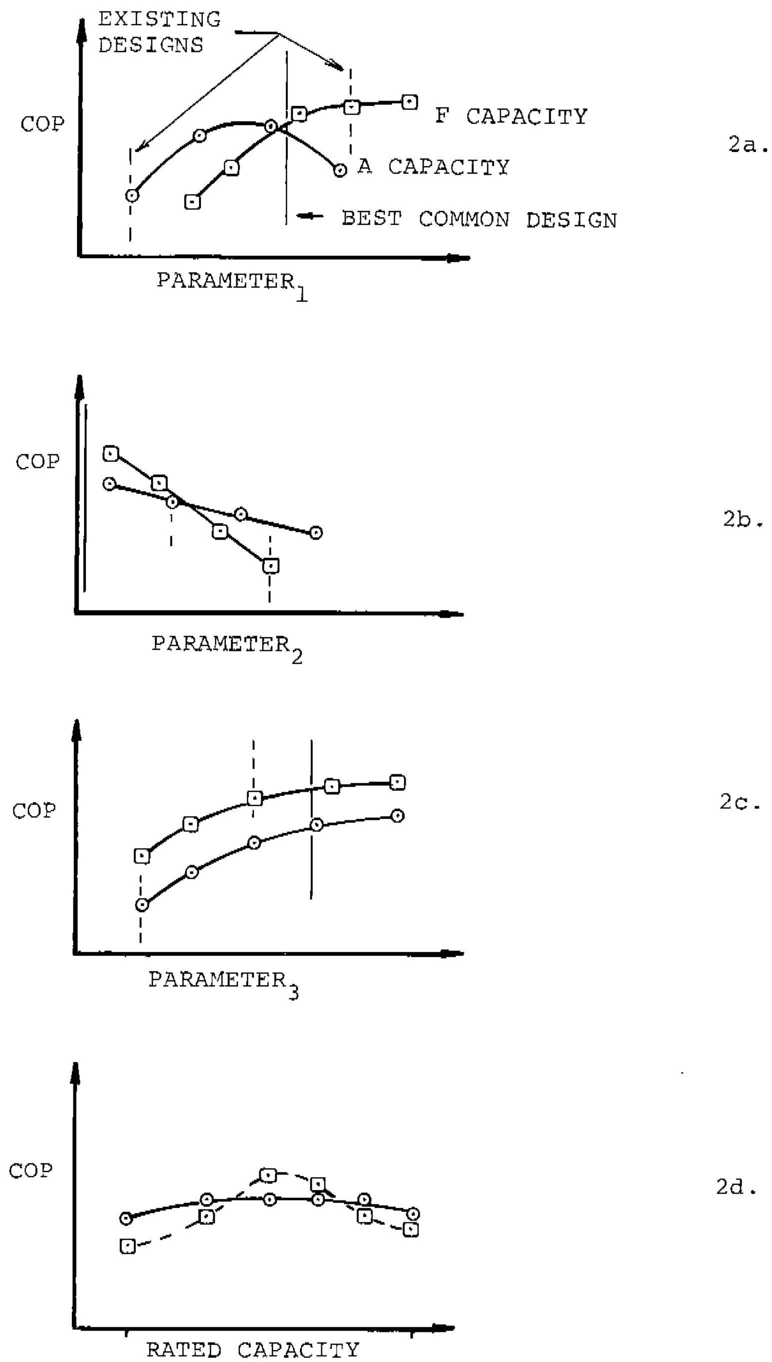
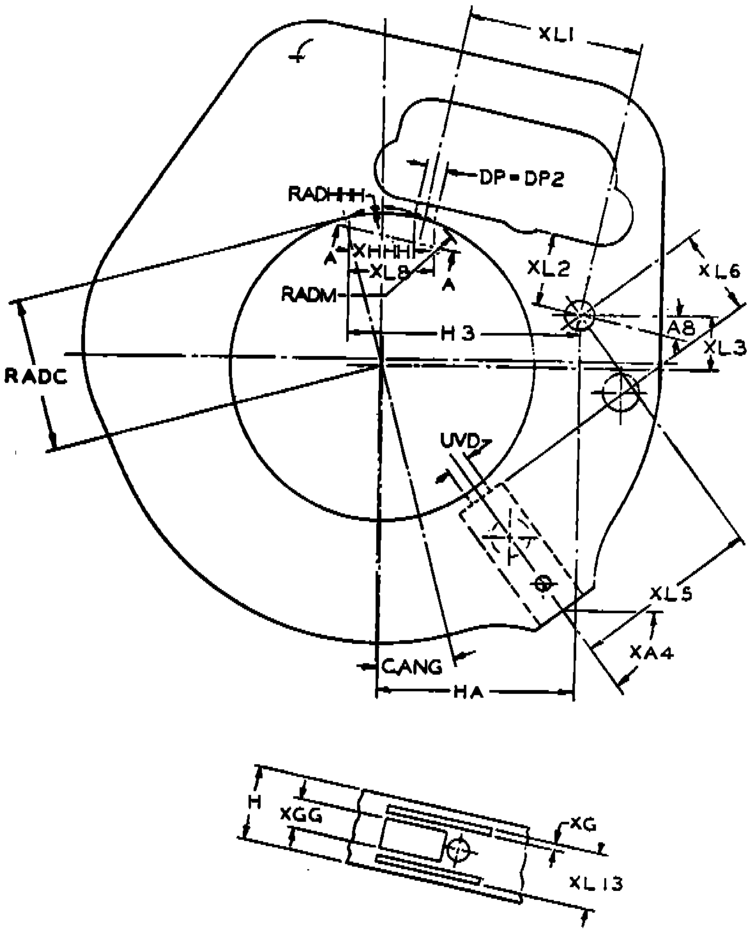


FIGURE 2 - TYPICAL EXAMPLES SHOWING METHOD USED IN STUDY



VIEW A—A

FIGURE 3 - GENERAL SHAPE OF COMPRESSOR CYLINDER USED IN STUDY WITH GEOMETRIC PARAMETERS IDENTIFIED

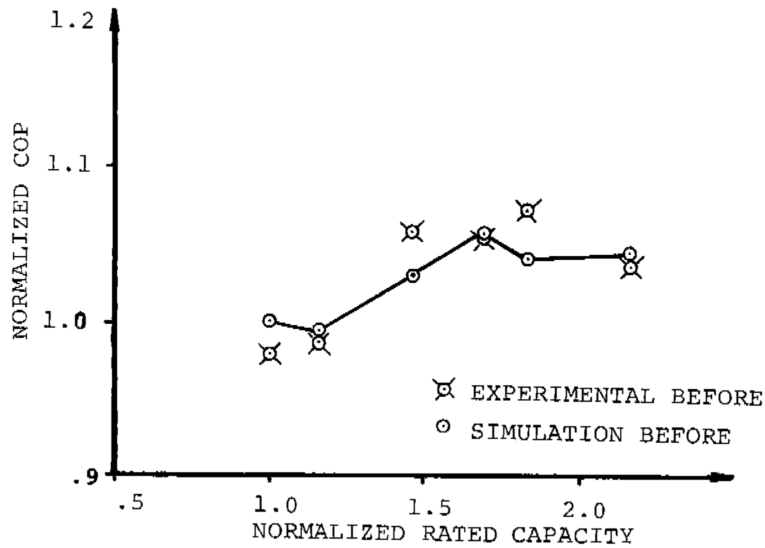


FIGURE 4 - COMPARISON OF THE EXPERIMENTAL AND THEORETICAL NORMALIZED COMPRESSOR PERFORMANCE BEFORE COMMONIZATION AND PERFORMANCE IMPROVEMENT

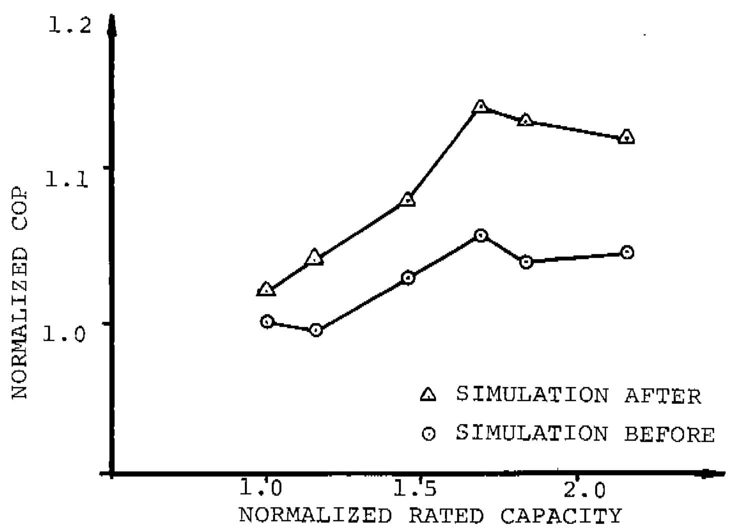


FIGURE 5 - COMPARISON OF THE THEORETICAL (SIMULATED) COMPRESSOR PERFORMANCE BEFORE AND AFTER COMMONIZATION AND PERFORMANCE IMPROVEMENT